



(11) Publication number : **0 528 638 A1**

(12)

EUROPEAN PATENT APPLICATION

(21) Application number : 92307406.6

(51) Int. Cl.⁵ : **F16D 59/00, F16D 67/02**

(22) Date of filing : 13.08.92

(30) Priority : 15.08.91 AU 7791/91
15.08.91 AU 7792/91
03.06.92 AU 2761/92

(43) Date of publication of application :
24.02.93 Bulletin 93/08

(84) Designated Contracting States :
AT BE CH DE DK ES FR GB GR IT LI LU MC NL
PT SE

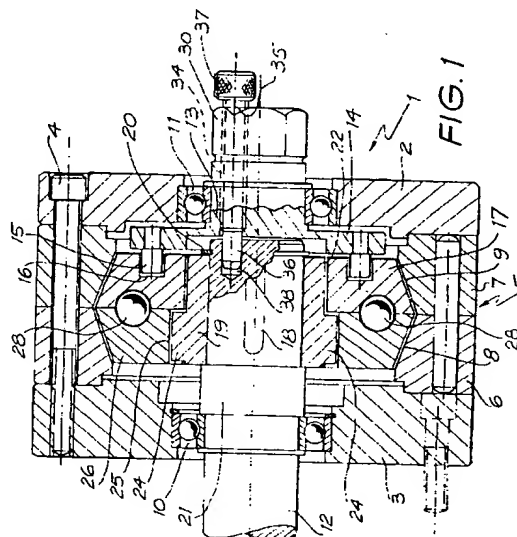
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(54) **Rotation check mechanism.**

(57) A cylindrical oil-filled casing (2,3,5) has an input shaft (12) on which are mounted a pair of braking rings (17,26) which are allowed limited axial and circumferential movement with respect to one another. The ring (26) is driven by the shaft and drives the other ring (17) through a circle of angularly-spaced, spring-biased balls (28) mounted in tear-drop shaped recesses in the opposed faces of the rings (17, 26). Each ring has a frusto-conical braking rim extending parallel to a frusto-conical braking surface (8,9) provided on the casing part (5). The rims of the rings (17,26) have angularly-spaced, shallow, chordal segments removed to provide recesses which, during rotation of the shaft (12), pump oil between the braking surfaces of the rings and the casing, to form hydraulic cushions therebetween. Reversal of the shaft torque causes the balls (28) to force the rings (17,26) apart and into braking engagement with the casing, to check rotation of the shaft. The pumping action of the recesses diminishes during braking so that the cushions collapse and do not impair braking.



Field of the invention

This invention relates to mechanisms for abruptly checking motion of a rotary drive, and is more specifically concerned with check coupling mechanisms and in-line braking mechanisms.

State of the art

In United States Patent No. 5,090,529, hereby inserted by way of reference, is described a check coupling mechanism constructed as a backstop brake. The mechanism described in Figure 4 of said Patent comprises an oil-filled casing having collinear input and output shafts projecting, respectively, from opposite sides. A gear pump driven by the output shaft is provided in the casing to feed oil under pressure into frusto-conical gaps located between complementary braking surfaces disposed in the chamber. Each of the shafts extends into the chamber and supports within it a rotary ring having a frusto-conical rim disposed adjacent one of the braking surfaces of the casing. Springs are located between the rings as are thrust-transmitting means provided by balls and tear-shaped recesses provided in opposed faces of the rings. The thrust-transmitting means serve to transfer torque from the ring mounted on the drive shaft, to the ring mounted on the driven shaft so that both rings rotate in the same direction. The rings are allowed to move axially and angularly with respect to one another. The recesses in the rings provide camming surfaces which, when the torque is applied in one direction between the rings, causes the balls to locate within the deepest portions of the recesses so that the frusto-conical rims of the rings are spaced from the braking surfaces in the casing. Oil pressure from the gear pump is applied to the frusto-conical gaps between the ring rims and the complementary braking surfaces, to hold them spaced from one another so that the drag of the mechanism on the drive shaft is limited only by the viscosity of the oil cushion provided in the frusto-conical annuli located between the braking surfaces of the rings and the casing. The springs located between the rings serve to urge them apart so that if the oil pressure in the cushions diminishes, the springs can thrust the two rings apart and into braking engagement with the casing.

If the direction of torque transmission from the input shaft to the output shaft of the mechanism is reversed, the balls run up the floors of the tear-shaped recesses which act as cams, and force the two rings apart and into braking engagement with the casing. The driven shaft then slows down rapidly and the oil pressure provided by the gear pump diminishes so that there is a collapse of oil pressure in the hydraulic cushions between the braking surfaces. Braking thus becomes accumulative and rapidly brings the drive shaft to a halt. The above-described arrangement is

effective in operation but the provision of the gear pump and ducting in the casing naturally tends to increase the cost of the mechanism.

Summary of the invention

In accordance with the present invention, a mechanism for abruptly checking the motion of a rotating input shaft, has an oil-filled casing containing two rings which are axially and angularly displaceable with respect to one another; braking surfaces provided on respective rings and which lie on respective frusto-conical surfaces; opposed braking surfaces provided on the casing and facing those of the rings, the braking surfaces of the casing also lying on frusto-conical surfaces which are complementary and parallel to those of the rings; resilient means acting to urge the two rings apart; and, a thrust-transmitting assembly composed of balls located in pockets and camming surfaces lying between the opposed faces of the rings, the thrust-transmitting means responding to a reversal in the direction of a driving torque applied to one of the rings by forcing the rings apart so that their rotation is abruptly checked by engagement of their braking surfaces with the braking surfaces of the casing; in which mechanism one of the braking surfaces has a series of circumferentially-spaced recesses dividing it into spaced arcuate braking zones, and each recess converges towards at least one of the braking zones flanking it. The effect of the recess converging towards a flanking zone is to exert a pumping action driving the recess oil into the annular gap between the opposed braking surfaces so that a cushion of oil under pressure is created between them to hold them apart.

Preferred features of the invention

The recesses are preferably formed by removing equi-spaced chordal segments from the rim of the rings. Alternatively, scalloped depressions may be provided in the braking surfaces of the casing to achieve the same effect. The creation of the hydraulic oil cushions between the braking surfaces is described in the aforesaid United States Patent. However the present invention creates the cushions more simply and cheaply. Further advantages achieved are that regions of oil shear stress between the opposed braking surfaces, when the mechanism is in its non-operating or quiescent mode, is substantially limited to the arcuately spaced regions of the gap lying between the opposed braking surfaces of the casing and the rings. The aggregated area of these regions is less than is the case with the arrangement described in the aforesaid United States Patent, and thus the heating of the oil is diminished also. Although there is a corresponding reduction in the effective areas of the braking surfaces of the mechanism, the braking

required can nevertheless be maintained by increasing the thrust which forces the braking surfaces towards one another when the mechanism assumes its operating mode, that is to say when it is exerting a checking function on the input drive, and also by the design and nature of the braking surfaces themselves. A further advantage is that each arcuately-spaced zone in which braking occurs, which actually participates in the braking, forms one face of a gap which has a circumferential aperture through which the oil cushion can dissipate when the mechanism switches from its quiescent mode to its operating mode. The response of the mechanism is thus quicker than with the arrangement used in the aforesaid patent, the aperture through which the oil pressure can dissipate in said Patent is effectively limited to the two annular ends of the gap between the braking surfaces.

Preferably, the resilient means used to urge the two rings apart comprises a set of compression springs arranged in inclined bores in at least one of the rings and which respectively urge the balls in a direction which forces the rings apart. As long as the mechanism is in its quiescent mode, the drag of the driven ring on the drive ring is sufficient to overcome the thrusts of the springs so that the balls remain in the deepest portions of tear-shaped pockets providing the camming surfaces. However, if the direction of torque is reversed, the drag direction is also reversed and this allows the thrusts of the springs to force the balls up the camming surfaces so that the two rings are rapidly driven apart. A relatively rapid response to a reversal in torque is then obtained. Such an arrangement avoids the use of separate springs between the rings, and also the risk that some only of the balls will respond to a torque reversal by driving the rings apart. In prior art arrangements, when the balls are not individually biased by their own springs but rely only on the relative angular movement of the two rings, it was not unusual for the rings to be driven apart by a few only of the balls with the result that those balls were subjected to excessive loading for which they were not designed. The result of the excessive loading is a slowing down of the response time of the mechanism, a tendency for it to judder badly when assuming its operating mode, and, in one or two cases, for the mechanism to fail under the excessive forces generated.

Introduction to the drawings

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which :

In the drawings

Figure 1 shows, in vertical section, a mechanism

for checking rotation of a shaft, shown partly broken away, when switching from its quiescent mode to its operating mode ;

Figure 2 is a scrap end view of the central part of the mechanism shown in Figure 1 as seen from the right hand end of that Figure;

Figure 3 diagrammatically represents an arrangement for balancing torque between two mechanisms of the form shown in Figure 1, operating in parallel;

Figure 4 is a diagrammatic end elevation of the peripheral margin of a rotary ring having an interrupted frusto-conical braking surface formed on its rim;

Figure 5 is a side elevation of the ring of Figure 4;

Figure 6 is a view of the ring of Figure 5 as seen from the right hand side of the Figure and illustrates a ring of tear-shaped recesses or pockets provided in the ring and containing the respective spring-biased balls forming thrust-transmitting means;

Figure 7 is a vertical section through two rings arranged as shown in Figure 1, and taken along the line and in the direction indicated by the arrows VII-VII in Figure 6;

Figure 8 is an enlarged view of the central portion of Figure 7 to show the arrangement of one ball and its biasing spring in more detail;

Figure 9 is a vertical section showing principal parts of a mechanism constructed as a uni-directional drive coupling with a reverse uni-directional check;

Figure 10 is a scrap section taken on the line and in the direction indicated by the arrows X-X in Figure 9;

Figure 11 is a view corresponding to Figure 9 but showing the principal parts of a coupling providing a bi-directional drive and a reverse bi-directional check;

Figure 12 is a scrap section taken on the line and in the direction indicated by the arrows XII-XII in Figure 11;

Figure 13 is a view corresponding to Figure 9 but showing the principal parts of a mechanism which provides a check coupling having a bi-directional drive and a reverse uni-directional check; and

Figure 14 is a scrap section taken on the line and in the direction indicated by the arrows XIV-XIV in Figure 13.

First embodiment description

The check mechanism shown in Figures 1 and 2 is of the general type described with reference to Figure 3 of aforesaid US-A-5,090,529 incorporated here by way of reference. It has a casing 1 provided with

opposite end caps 2 and 3 which are attached by bolts 4 to respective ends of an intermediate cylindrical section 5 formed from two annular members which provide a pair of opposed frusto-conical braking surfaces 8, 9 inside the casing 1.

The end caps 2, 3 have respective central bearings 10, 11. The bearing 10 supports an input drive shaft 12 to which is applied the torque to be monitored by the check mechanism, and the bearing 11 supports a stub shaft 13 of a rotary spider frame assembly 14. The assembly 14 carries a ring of release pins 15 which locate in respective pockets 16 provided in a braking ring 17. This has a generally frusto-conical braking rim shown in Figures 4 and 5 and which is forced, by axial displacement of the ring 17, against the braking surface 9 when the check mechanism is operated.

The shaft 12 is connected by a key 18 to a surrounding stepped collar 19 inside the casing 1. The collar 19 is held against axial displacement by a circlip 20 on the shaft 12 at one end, and by a shouldered portion 21 of the shaft at the other end. The collar 19 has a cylindrical end portion 22 providing a bearing for the ring 17, and its other end portion is provided with two arcuate and circumferentially-spaced radial flanges 24. The flanges 24 respectively fit loosely into arcuate slots 25 provided on the inside face of a second braking ring 26. The engagement of the flanges 24 in their respective slots 25 allows the ring 26 limited axial and circumferential movement on the shaft 12.

Figures 4 and 5 show the generally frusto-conical rim portion of the ring 26 in more detail. The other ring 17 is similarly constructed. The ring 26 has its frusto-conical rim of interrupted form, in that it is divided into arcuately-spaced braking zones 52 by the formation of a circumferentially-spaced chordal flats 51. These flats provide pumping recesses and reduce the surface area of the ring rim dedicated to braking, to between 40% and 44% of the area available for braking when the opposed braking surfaces are truly frusto-conical as is the case with the aforesaid United States Patent. However this disadvantage is capable of being entirely offset by an appropriate choice of braking surfaces materials and by the aggregated thrusts of springs which force the two rings apart to initiate braking when the mechanism switches from its quiescent mode to its operating mode.

Located between the two rings 17 and 26, at angularly spaced positions, are thrust transmitting means shown in more detail in Figures 6, 7 and 8. Such means are provided by a set of arcuately spaced steel balls 28 held partly in each of two tear-shaped recesses or pockets provided in the opposed faces of the rings 17 and 26. The tails of the tear-drops extend in one direction of rotation on one of the rings, and in the other direction of rotation on the other ring. The floor of each recess slopes upwards towards

its 'tail' to provide a camming surface and each of the balls is provided with its own bias spring 56 located in an inclined bore 55 and which urges the ball up the camming surface of the associated recess. The two rings 17 and 26 are thus urged apart by the aggregated spring bias provided on the balls 28. The torque of the shaft 12 applied to the ring 26 by way of the flange 24 and the end-wall of the slot 25, angularly displaces the ring 26 with respect to the ring 17 in a direction which forces the balls 28 against the resilience of their springs to the positions shown in Figure 7. In these positions illustrated, the balls are located in the deepest regions of the recesses or pockets. This corresponds to the quiescent mode of operation of the mechanism. The bore 55 is inclined at an angle to the direction of rotation of the ring 17 in which it is formed as illustrated.

The stub shaft 13 protrudes out of the end cap 2 to provide a release facility. It is formed at its end with a hexagonal nut 30 to which a spanner (not shown) can be applied progressively to allow the shaft 12 to be turned, under complete manual control, in the direction of the reversed torque, after the check mechanism has assumed its operating mode. This is described in more detail in the aforesaid United States Patent. A further bore 34 which is eccentric to the axis of the stub shaft 13, extends inwardly through the stub shaft and contains a latching screw 35 having an axial pin 36 at one end inside the casing, and a knurled opposite end 37 to enable the axial position of the screw 35 to be altered from outside of the casing 1. The end face of the shaft 12 inside the casing 1 is provided, off-centre, with a well 38 at a position at which it can be engaged by the pin 36 when the screw 35 is screwed into the bore 34. During normal operation of the mechanism when it is in its quiescent mode, the screw 35 is unscrewed sufficiently to disengage the pin 36 from the well 38.

Operation of the first embodiment

During quiescent operation of the mechanism shown in Figure 1, the two rings 17 and 26 assume the positions shown in Figure 7 at which the balls 28 are contained in the deepest portions of the tear-shaped recesses 53 and 54 on respective rings 17 and 26. There is then minimum drag applied to the shaft 12. The springs 56 are held in compression by the drag of the ring 17 on the ring 26. This drag is produced by the viscosity of the oil between the braking zones 52 of the ring 17 and the frusto-conical braking surfaces 8 and 9 formed inside the casing 1.

The rotation of the rings 17 and 26 and the shapes of the recesses 51, causes the oil in the recesses 51 to be wedged by their pumping action into the gaps formed between the braking zones 52 of the rings and the opposed frusto-conical surfaces of the casing. Such pumping action creates cushions of oil

under pressure between the braking surfaces of the rings and casing so that they are held apart and do not rub against one another. The oil in these gaps is subjected to shear stress but it is prevented from overheating by flowing out of the gaps via the annulus formed around the edges of the zones 52, so that there is a relatively rapid circulation of oil through the pressure cushions. The degradation of the oil through the effects of the shear stress and heating, is therefore relatively small.

If the drive torque on the shaft 12 is removed, the combined thrust of the springs 56 acting on the balls 28 forces the two rings 17 and 26 apart against the pressure of the oil cushions. The resistance to this thrust rapidly diminishes as the oil flows quickly out of the cushions when the pumping effect of the chordal recesses diminishes as a result of the speed of rotation of the rings 26, 17 diminishing. Also, as the balls are acted on by their individual springs, there is no risk of some only of the balls being effective to force the rings apart. Thus a prompt reliable and abrupt braking action ensues with the thrust-transmitting means provided by the balls 28 and the camming surfaces of the associated recesses all being effective.

When the mechanism is in its operating mode and the shaft 12 has been checked to a halt, the stub shaft 13 protruding out of the end cap 2 can be used to provide a release facility, as fully described in the aforesaid United States Patent. However there are situations when it is required temporarily to neutralise the checking action of the mechanism when a reverse torque is applied, for example, to enable maintenance routines to be carried out. This is effected as follows.

With the shaft 12 manually held against rotation, the nut 30 is rotated sufficiently in a direction which moves the balls 28 into the deepest portions of their recesses so that the two rings 26 and 17 are at their closest relative positions. This simultaneously brings the pin 36 opposite the well 38. The knurled head 37 may then be turned manually to engage the pin 36 with the well 38. This locks the angular position of the shaft 12 and thus the ring 26, with that of the spider assembly 14 and thus the ring 17. The balls 28 are then prevented from moving out of their deepest positions in their recesses and the braking capability of the mechanism is temporarily neutralised so that the shaft 12 can be turned freely in either direction to enable the required maintenance to be carried out. On completion of the maintenance, the nut 37 is unscrewed to free the pin 36 from the well 38. The check mechanism can then function normally.

Second embodiment description

Referring to Figure 9, a support plate 101 carries two coaxially mounted annular members 102, 103 at-

tached by bolts 104 to provide a casing for the coupling. The members 102, 103 each provide an internal frusto-conical braking surface. The braking surface of the member 102 is referenced 105 and that of the member 103 is referenced 106. The two braking surfaces are adjacent and opposed to one another, as illustrated.

Shaft 107 passes concentrically through the two members 102, 103 and carries a collar 108 between them. The collar 108 has an internal key-way 109 which receives a key attaching the collar stiffly to the shaft.

The collar 108 has its external surface providing two axially-spaced circumferential zones 110 and 111. The zone 110 is equipped with four equiangularly spaced dogs 112 each of rectangular cross-section, and the zone 111 has a smooth cylindrical external surface.

A ring 114 having a generally frusto-conical rim 115 formed with chordal flats as has already been described with reference to Figures 4 and 5, is internally provided with slots 144 each of which accommodates one of the dogs 112 so that the ring 114 is slidable and loosely mounted on the dogs 112. This permits its rim 115 to have limited angular, radial and axial displacement with respect to the member 103. Rim 115 is arranged opposite and substantially parallel to the braking surface 106, and is axially spaced from it when the coupling is in its quiescent mode.

A second ring 116 is mounted on the cylindrical zone 111 of the collar 108 and is loosely coupled to a coaxially arranged output shaft 146 by a ring of studs 140 having their shanks attached at angularly spaced positions 141 to a radial flange 142 on the shaft, and their heads 143 located in holes 117 in the member 116.

Mounted between the opposed and spaced faces of the rings 114 and 116 are a set of incompressible thrust-transmitting means 119. Each such means comprises a spring biased steel ball 120, identical to the ball 28 of Figure 1, having diametrically opposite parts of its surface seated in two opposed tear-shaped recesses 121, 122 provided in the rings 114 and 116 as has already been described with reference to Figures 6, 7 and 8. The recesses each provide a camming or ramp surface extending circumferentially with respect to the axis of rotation of the rings and the camming surfaces of each pair of opposed recesses respectively extend in opposite directions as shown in Figure 10. Thus relative angular movement between the rings in one direction of rotation, causes each ball to seat in the deepest parts of its flanking recesses, whereas relative angular movement of the rings in the opposite direction causes the balls to travel up the camming surfaces towards the shallowest parts of the recesses with the result that the rings 114, 116 are thrust apart by the thrusts of the individual springs on the balls 120.

Operation of second embodiment

In operation of the coupling described, rotational drive of the shaft causes rotation of the collar 108. The dogs 112 are carried into engagement with one of the radially extending sides of the slots 144 in the ring 114 causing it to rotate. This rotates the ring in a direction which moves the balls 120 into the deepest parts of their recesses and they consequently impart the rotational drive to the other ring 116 as the coupling, at this time, is in its disengaged or quiescent mode. The ring 116 is free to rotate on the zone 111 of the collar 108 and thus transmits drive to the output shaft 146.

If the input drive to the coupling ceases, it assumes its operative or engaged mode at which it checks the load. This results from the ring 114 moving angularly with respect to the ring 116 in a direction which causes the balls 120 to ride up their camming surfaces and force the two rings apart. This is permitted by the slots 144 riding lengthwise along the dogs 112. The interrupted frusto-conical rims of the rings move apart and into engagement with the braking surfaces of the casing. The full torque of the load is applied to the thrust-transmitting means so that braking of the rotation of the input and output shafts of the coupling occurs abruptly. As both rings 114, 116 are allowed limited radial movement on the collar, the effect of their rims engaging the braking surfaces 106, 105 is to produce a self-centring effect so that the areas between which braking occurs is maximised.

To release the coupling once it has engaged, it is necessary to turn one of the two shafts in a direction which causes the balls 120 to ride down their camming surfaces into the deepest parts of their associated recesses 121, 122 once again. This occurs automatically when the drive is re-applied to the ring 114 or when the ring 116 is manually rotated in the appropriate direction by the application of a spanner (not shown) to a hexagonal nut 150 provided around the outer end of the shaft 146.

As with all of the other embodiments described in this specification, the casing is oil-filled, and the balls 120 are spring biased, as described in Figures 6, 7 and 8 above.

Third embodiment description

The check coupling provided by the arrangement of Figure 11 provides a bi-directional drive with a bi-directional check being applied if the drive is disconnected in either direction. Parts of Figure 11 corresponding to those already described with reference to Figure 9 are similarly referenced but are in the '200' series and will not therefore be again described to save unnecessary repetition of description.

The collar 208 of Figure 10 has the dogs 212 extending axially throughout its length. A second collar

230 of the same internal and external dimensions as collar 208 is arranged in-line with the collar 208 and is likewise equipped with four external dogs 231 of the same cross-sectional shape as the dogs 212, and aligned therewith. The collar 230 has an internal keyway 232 for receiving a key which attaches it stiffly to an output drive shaft 233.

The rings 214, 216 have their opposed faces provided with double-ended tear-shaped recesses 235, 236 as shown in Figure 12. When the two rings 214, 216 are at their closest positions to one another, as illustrated in Figure 11, the balls 220 occupy central positions in their double-ended recesses 235, 236. Each of the balls is provided with associated bores containing compression springs (not shown) arranged as shown in Figures 7 and 8, so that when the coupling is to switch from its quiescent mode to its operating mode, the springs exert a force on the balls causing the two rings 214, 216 to be forced apart thus bringing their frusto-conical rims 215, 218 into engagement with the frusto-conical braking surfaces 206, 205 of the casing. The rings 214, 216 once again have frusto-conical rims formed with pumping recesses (see Figures 4 and 5.)

It will be noticed that in the arrangement of Figure 11, the ring 216 is axially longer than the second collar 230 and is provided internally with axial slots 234 of the same cross-sectional shape as the slots 244 inside the ring 214 and which are aligned therewith.

The coupling is assembled with the input drive shaft 207 attached to the collar 208 and the output drive shaft 233 attached to the collar 230. The dogs 212 of the collar 208 are contained partly in the slots 244 of the ring 214 and partly in the slots 234 of the ring 216. The dogs 231 of the second collar 230 are contained wholly within the slots 234 of the ring 216 as shown.

Operation of third embodiment

The coupling of Figure 11 operates as follows: During starting up, when the coupling is in its quiescent mode, the rotation of the drive shaft 207 is transferred to the collar 208 via the keyway 209. The dogs 212 of the collar 208 transmit the drive to both rotary rings 214, 216 in unison, by way of the slots 244 and the balls 220. In all cases the slots 244 and 234 are substantially wider circumferentially, and slightly wider radially than the dogs they contain. The rotation of the ring 216 is transferred by the radially extending end walls of the slots 234 to the dogs 231 of the second collar 230 which is stiffly keyed to the output shaft 233 so that it rotates with the input shaft 207. If the output shaft 233 should accelerate beyond the speed of the input shaft 207, or if the drive to the input shaft 207 should fail, the dogs 231 will advance in phase with respect to the dogs 212 and cause angular movement of the ring 216 to occur with respect to

the ring 214 under the influence of the spring-biased balls 220.

The balls 220 then run up the camming or ramp surfaces at one end of the recesses 235, 236 and force the rings 214, 216 apart so that their rims 5 brake on the surfaces 206, 205 and both input and output shafts are brought to an abrupt halt. To enable rotation to re-commence, the relative angular movement between the rings 214, 216 must be reversed to allow the balls to re-assume their mid-position in the deepest parts of the recesses 235, 236 as shown in Figure 2.

Because the recesses are double-ended, the same abrupt braking will occur irrespective of the direction of rotation of the input shaft 207 driving the collar 208. The only difference will be that braking will occur as a result of the balls 220 running up the camming or ramp surfaces at the other pair of ends of the opposed recesses 235, 236.

Fourth embodiment description

The arrangement shown in Figure 13 provides a bi-directional drive with a reverse unidirectional check facility. Parts of Figure 11 corresponding to equivalent parts of Figure 9 are similarly referenced but in the '300' series. To avoid needless repetition, these parts will not be again described in detail.

The arrangement of coupling of Figure 13 is virtually the same as that shown in Figure 11 except that the tear-drop shaped recesses 321, 322 are single-ended as shown in Figure 14, and not double ended as shown in Figure 12. As a consequence, the abrupt halting of rotation of the coupling will only occur when the relative angular movement between the rings 314, 316 is in one direction. This occurs when the spring-biased balls 320 run up the ramp or camming surfaces at one end of the recesses 321, 322 and force the rings 314, 316 apart. Angular movement of the rings 314, 316 in the reverse direction is prevented by the balls 320 engaging the end walls of their associated recesses 321, 322 and simply transmitting the rotational drive between the two rings 314, 316 which each has an interrupted frusto-conical rim (see Figures 4 and 5).

Fifth embodiment description

Figure 3 shows how two torque checking mechanisms can be safely operated in parallel.

It is common practice to provide a piece of equipment being monitored, with two torque checking mechanisms so that the torque to be checked, which can be very large, is shared between them. However, in practice, two torque-checking mechanisms seldom operate in precisely the same way. One of the mechanisms is invariably subjected to the full torque to be checked, before the other one commences opera-

tion. The arrangement shown in Figure 3 is designed to overcome this difficulty.

Figure 3 shows a drive shaft 42 whose rotation is to be checked abruptly in the event of the torque being reversed. The drive shaft 42 carries a pinion 41 which is in mesh with a pair of gear wheels 40' and 40". These are arranged at diametrically opposite positions on the pinion 41.

Each of the gear wheels 40', 40" is connected to the input shaft of a check coupling of the form shown in Figure 1. Each check coupling has a casing 1', 1" which, instead of being anchored to a bed plate is fixed to one end of an arm 43', 43". The arm allows the casing limited rotational movement through a degree or two. The end of the arm lies between an abutment 44', 44" and a piston 45', 45". Each of the pistons 45', 45" has an associated cylinder 46', 46" interconnected by way of hydraulic piping 47.

Operation of fifth embodiment

During normal operation of the equipment when checking of the torque is not required, the pinion 41 rotates in the direction of the arrow illustrated. This motion is imparted to the two check couplings which are in their quiescent modes. Their input shafts 12', 12" rotate in the direction of the arrows illustrated, so that the two arms 43', 43" press against their corresponding pistons 45', 45".

In the event that the direction of torque of the pinion 41 reverses, a corresponding rotational torque is applied to the two gear wheels 40', 40". In the unlikely event that they operate truly simultaneously, their arms 43', 43" will simultaneously impact on the abutments 44', 44" so that the torque to be checked be divided equally between them.

If, as is invariably the case, one of the check mechanisms, say that associated with the arm 43' operates first, its arm 43' will impact on the abutment 44'. This frees the piston 45' so that hydraulic oil passes from the cylinder 46" to the cylinder 46' by way of the piping 47. The upward movement of the piston 45" thus permitted, causes the arm 43" of the casing 1" to turn in a direction which momentarily removes torque from the associated check mechanism, causing it immediately to switch from its quiescent mode to its operating mode. Both check mechanisms thus operate together to assume their operating mode despite their characteristics being different, so that the reverse torque of the pinion 41 is divided equally between the two check mechanisms.

Claims

1. A mechanism for abruptly checking the motion of a rotating input shaft, having an oil-filled casing (2,3,4,5) containing two rings (17,26) which are

- axially and angularly displaceable with respect to one another; braking surfaces provided on respective rims of the rings and which lie on respective frusto-conical surfaces; opposed braking surfaces (8,9) provided on the casing and facing those of the rings, the braking surfaces of the casing also lying on frusto-conical surfaces which are complementary and parallel to those of the rings (Figures 1, 9, 11, and 13); resilient means acting to urge the two rings (17,26) apart; and a thrust-transmitting assembly composed of balls (28,120,220,320) located in pockets (53,54) and camming surfaces lying between the opposed faces of the rings (17,26), the assembly responding to a reversal in the direction of a driving torque applied to one of the rings (26) by forcing the rings apart so that their rotation is abruptly checked by engagement of their braking surfaces with the braking surfaces of the casing; characterised in that one of the braking surfaces (Figures 4 and 5) has a series of circumferentially-spaced recesses (51) dividing it into spaced arcuate braking zones (52), and each recess (51) converges towards at least one of the braking zones (52) flanking it.
2. A mechanism as claimed in claim 1, characterised in that the recesses (51) are respectively formed on the rim of each ring (17,26) by a series of equispaced chordal segments removed from the rim.
 3. A mechanism as claimed in claim 1 or claim 2, characterised in that each of the balls (28,120,220,320) is biased by its own spring (56) in a direction which urges the rings (17,26) apart, the spring being located in a bore (55) formed in the ring (17) and exerting a thrust on the ball acting to move it towards the shallow end of a tear-shaped pocket (53,54,121,235-6,321,322) in which the ball is located.
 4. A mechanism as claimed in claim 3, characterised in that each pocket is double-ended (235,236) and has its deepest part in the centre of the pocket, each ball (220) being acted upon by two springs (56) each located in its own bore (55).
 5. A mechanism as claimed in any one of the preceding claims, characterised in that at least one of the rings (114) is loosely mounted on dogs (112) attached to a collar (108) fixed to the shaft (107), the rings (114,116) being angularly and axially displaceable with respect to one another and each of the rings being also permitted limited tilting movement about its centre.
 6. A mechanism as claimed in claim 5, characterised in that input and output shafts (107,146) extend respectively into opposite sides of the casing (2,3,4,5) and are collinear, drive from the input shaft (107) to the output shaft (146) being transmitted by way of the ring (114) mounted on the collar (108) of the input shaft (107), the thrust-transmitting assembly (120) located between the two rings (114,116), and the second ring (116) which is mounted loosely on a spider (142) attached to drive the output shaft (146).
 7. A mechanism as claimed in any one of the preceding claims, characterised by a releasable latching device (36,38) operable to latch the two rings (17,26) together when the balls (28) are in positions at which the braking effect of the mechanism is a minimum (see figure 1), so that the input shaft (12) can be turned freely in either direction.
 8. A mechanism as claimed in any one of the preceding claims, characterised in that its input shaft (12') is connected with that of a similar mechanism (1'') to a source of rotational drive (41) to be checked, the casings of the two mechanisms (1', 1'') being interconnected through load-balancing means (47,43',43'',45',45'',46',46'') which is effective to ensure simultaneous operation of both mechanisms (1',1'') despite their operating characteristics being slightly different.
 9. A mechanism as claimed in claim 8, characterised in that said load balancing means comprises two piston-and-cylinder units (45',45'',46',46'') interconnected by hydraulic piping (47), and arms (43',43'') attached to the casings of the mechanisms (1',1'') and operating respective units (45',45'',46',46'') in response to limited angular movements of the casings caused by the operation of the associated mechanisms (1',1'').

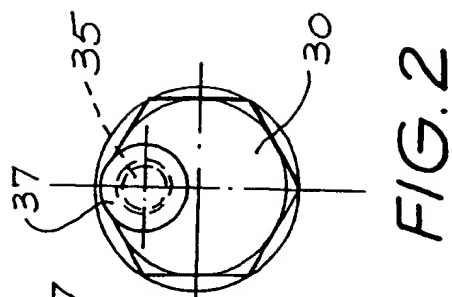
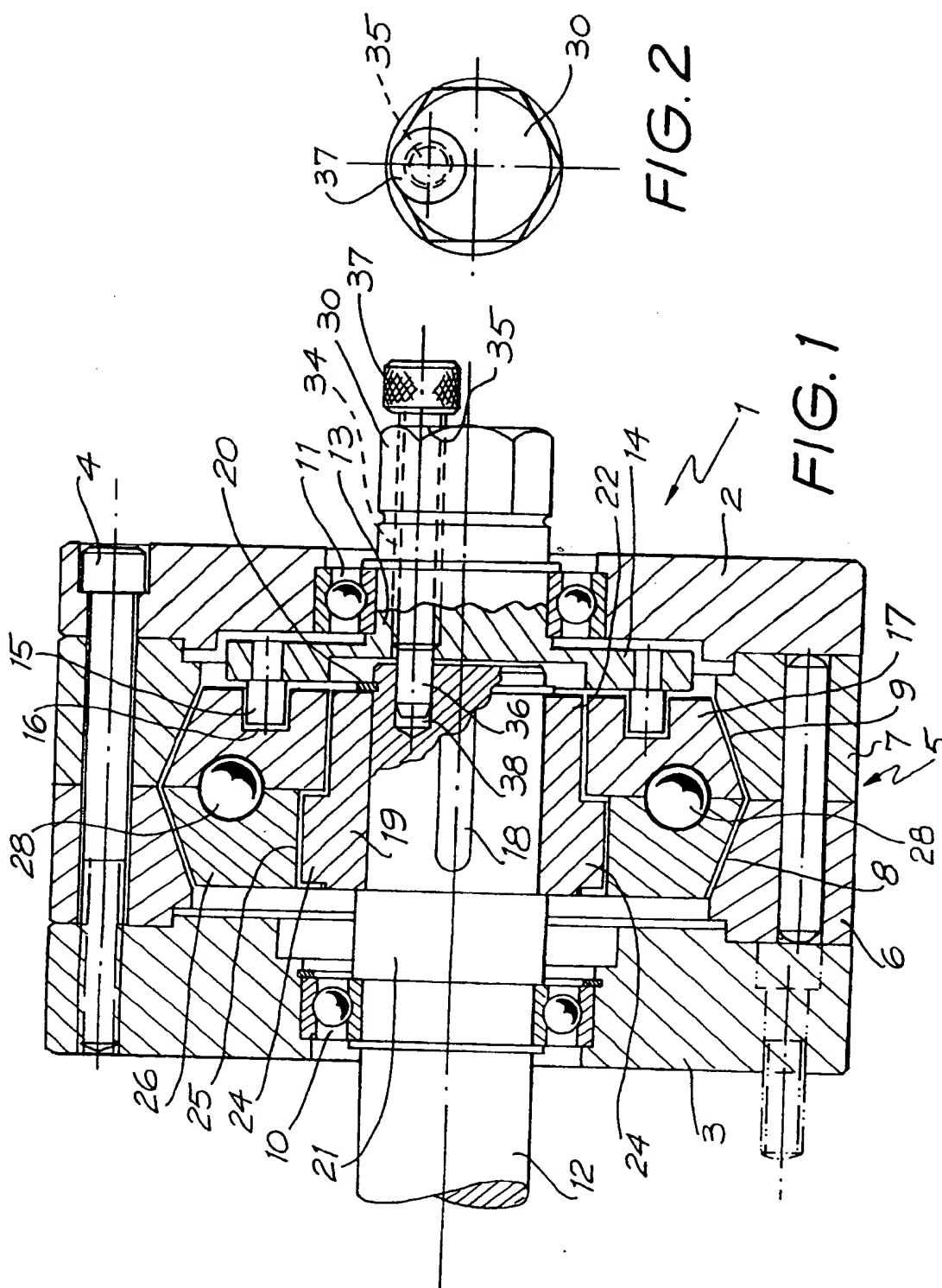
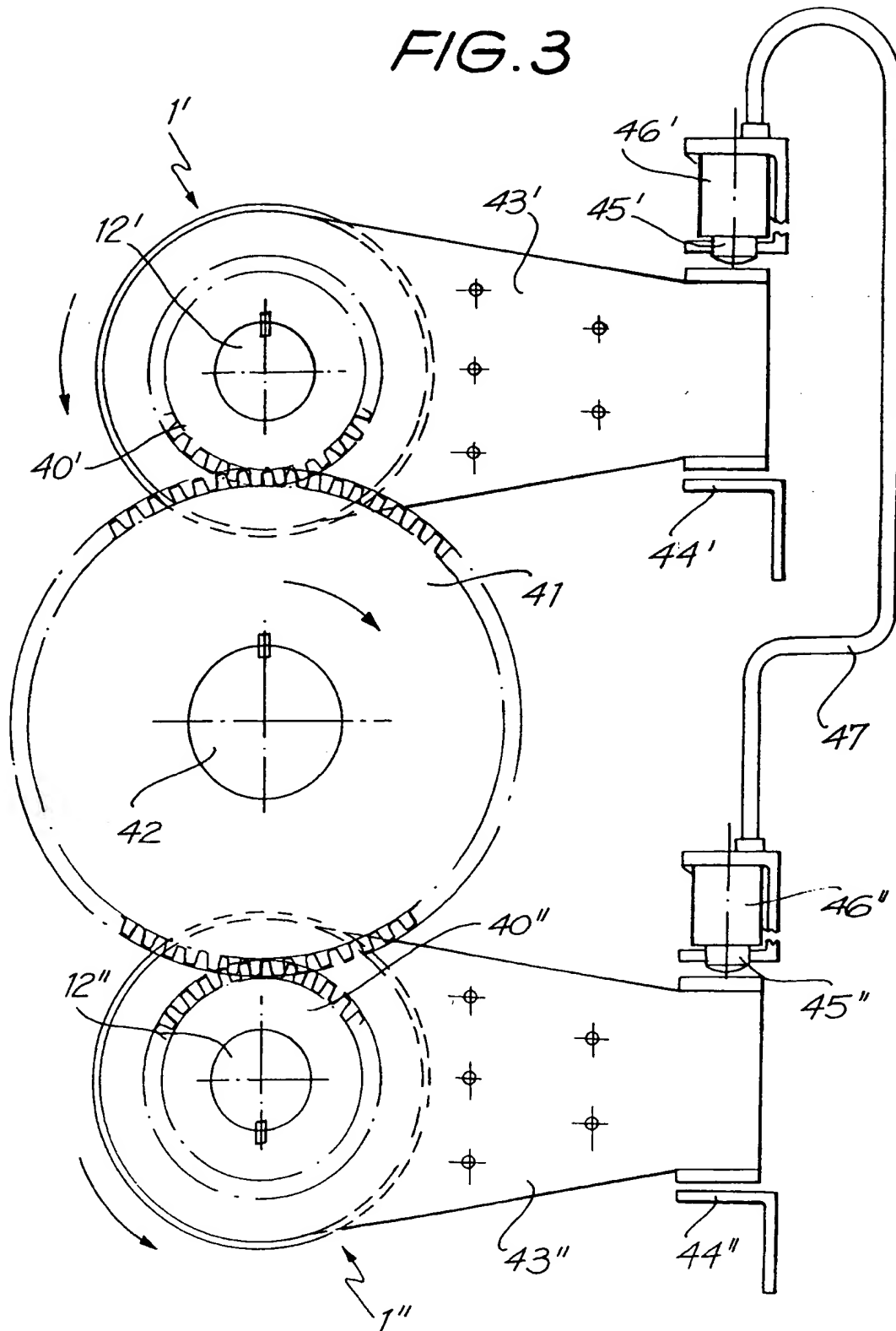
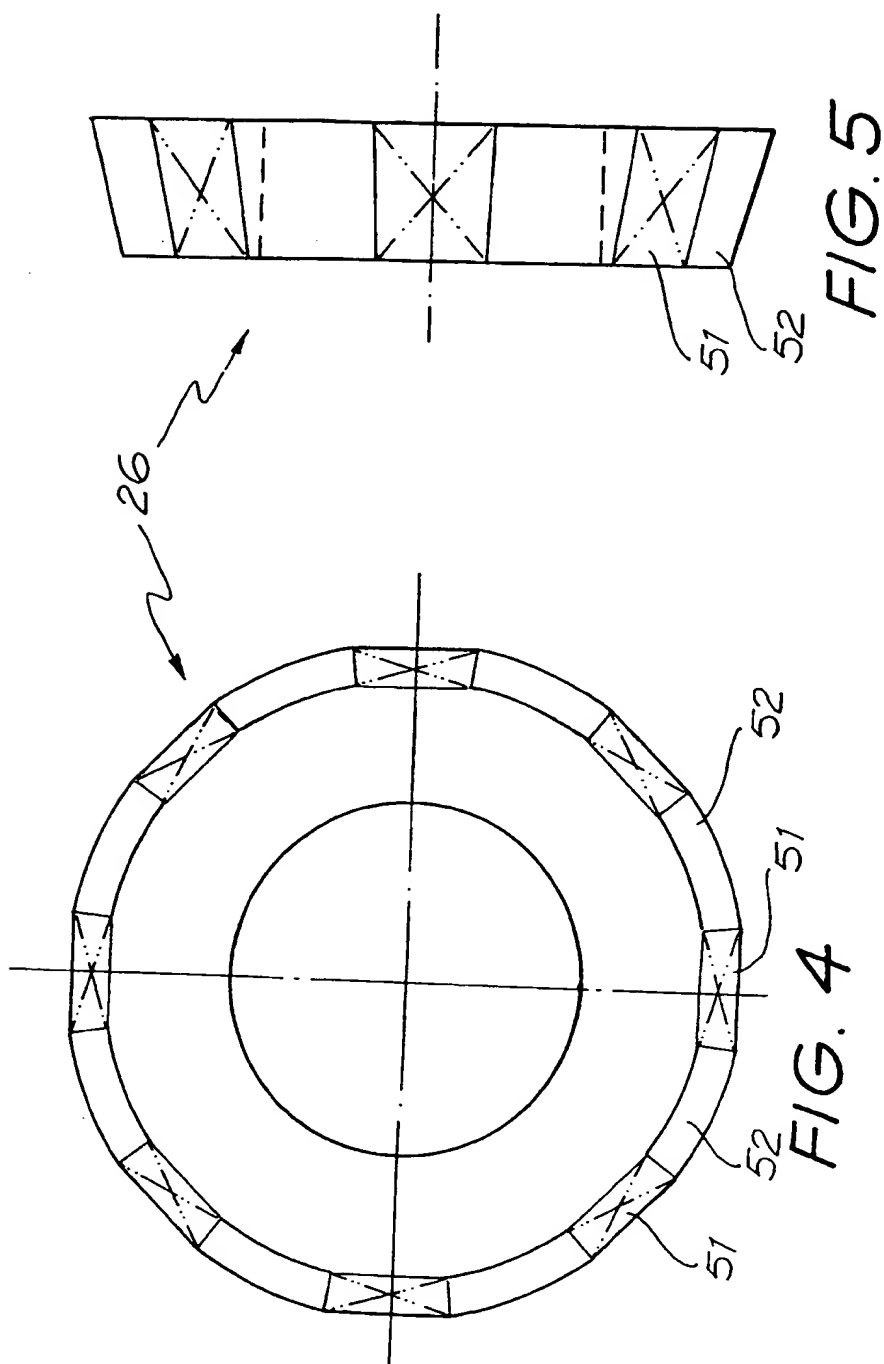
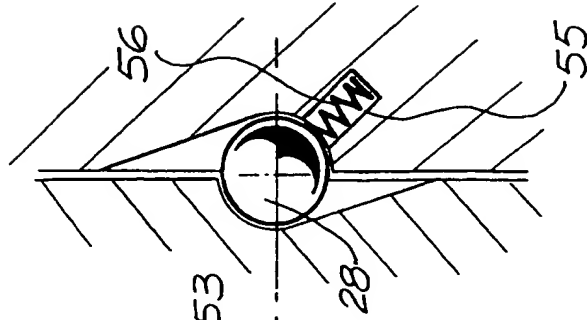
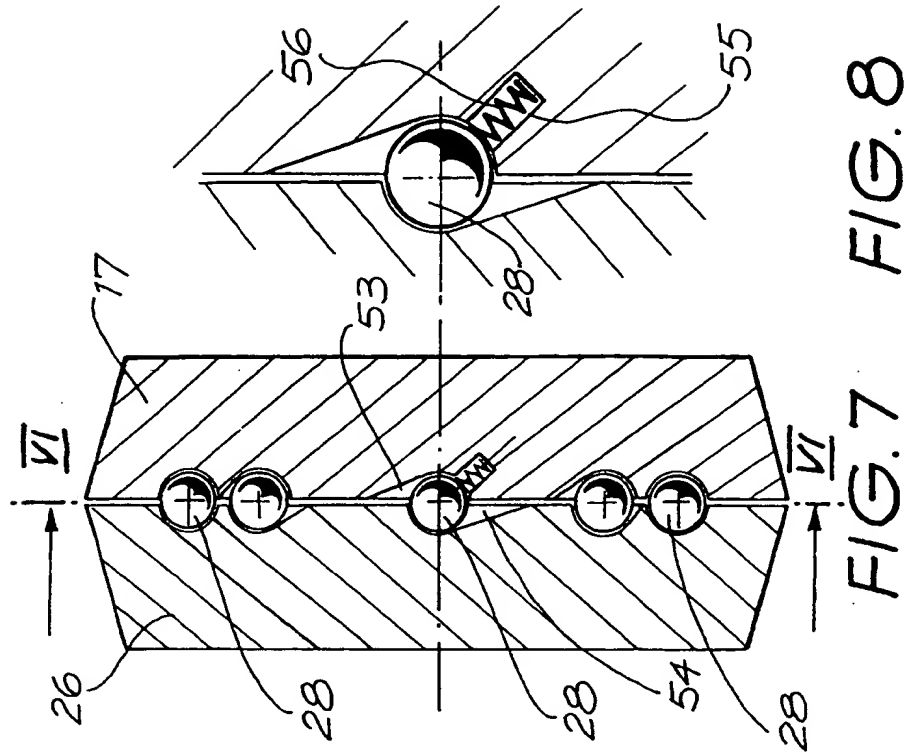
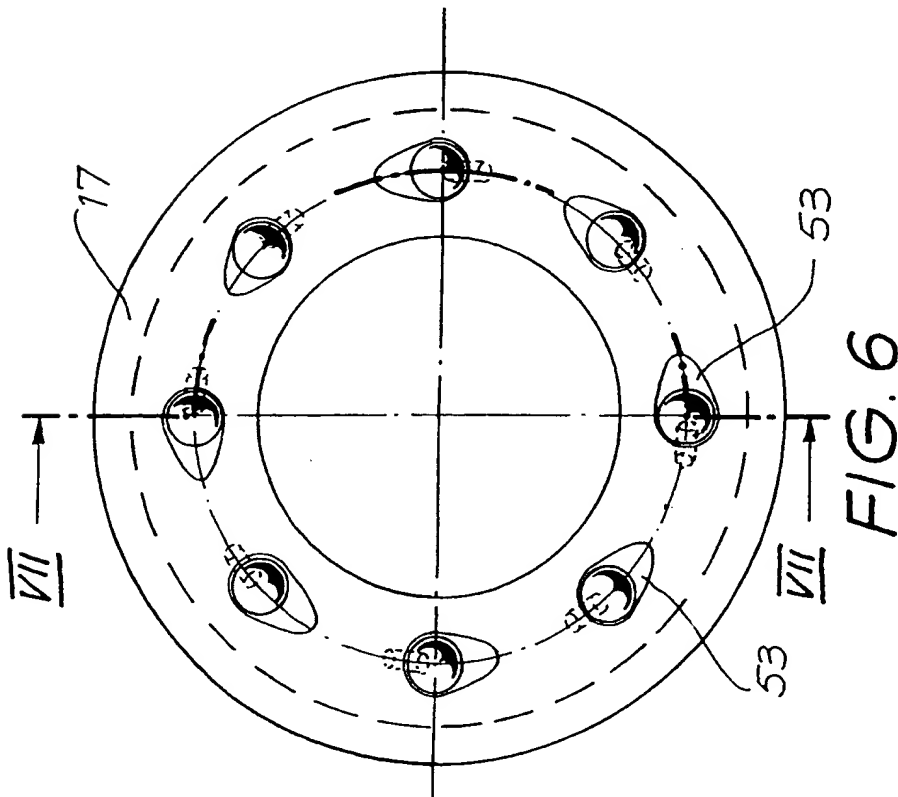
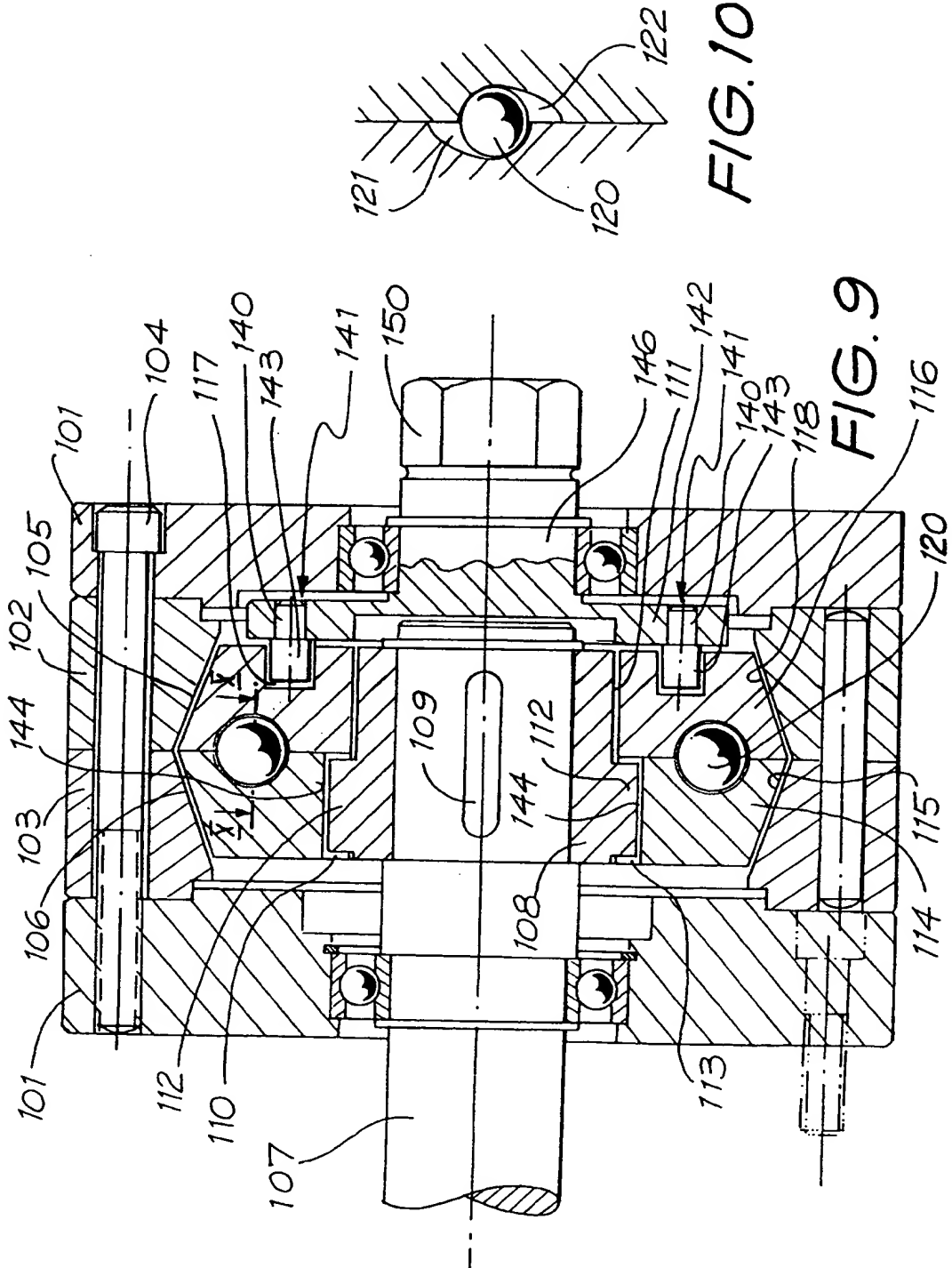


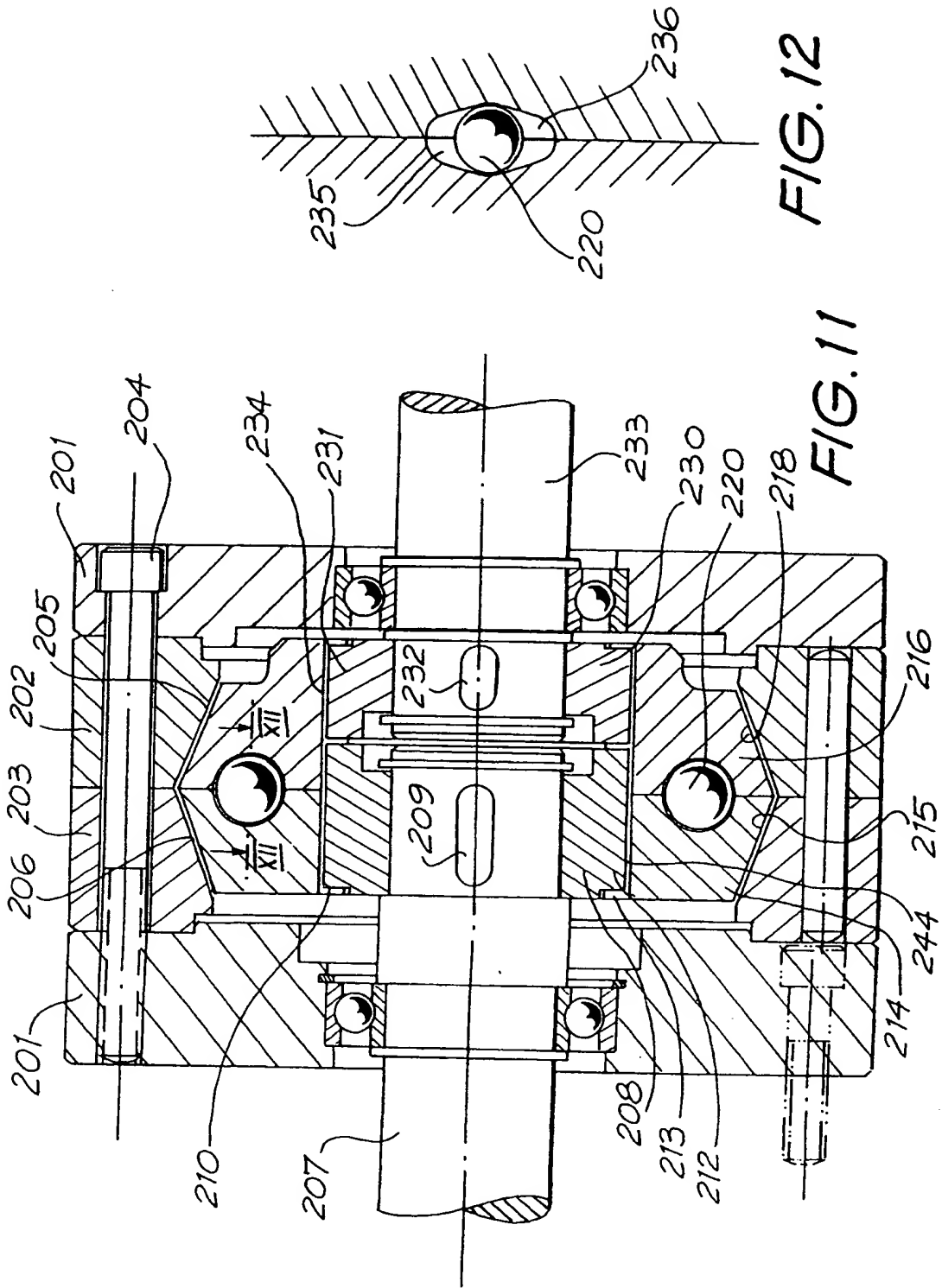
FIG. 3

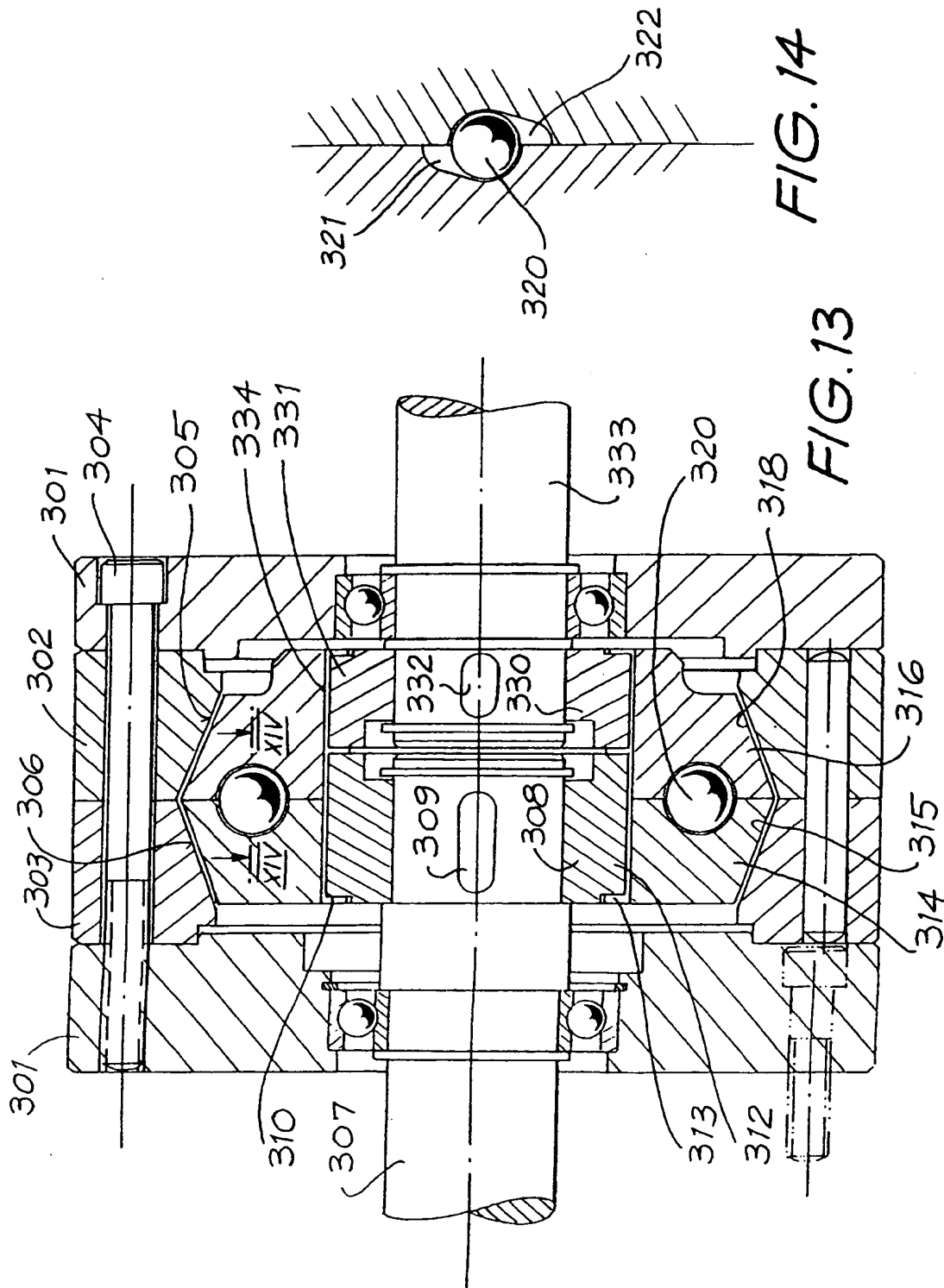














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EUROPEAN SEARCH REPORT

Application Number

DOCUMENTS CONSIDERED TO BE RELEVANT			EP 92307406.6
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
D, P	US - A - 5 090 529 (FAHY et al.) * Fig. 4 *	1	F 16 D 59/00 F 16 D 67/02
A	US - A - 4 909 363 (TROMMER) * Fig. 1-3 *	1	
A	GB - A - 2 237 855 (DANA CORPORATION) * Fig. 1-5 *	1	
A	GB - A - 2 041 472 (DUNLOP LIMITED) * Fig. 1-6 *	1	
P, A	US - A - 5 065 854 (SERRANO et al.) * Totality *	1	
			TECHNICAL FIELDS SEARCHED (Int. Cl.5)
			F 16 D 59/00 F 16 D 67/00
The present search report has been drawn up for all claims			
Place of search VIENNA	Date of completion of the search 28-10-1992	Examiner KAMMERER	
CATEGORY OF CITED DOCUMENTS		T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document	
X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document			

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